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Design and Analysis of a Spindle of Special Purpose Drilling Machine by using Solid Works

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Abstract

The growth of manufacturing industries largely depends on rate of production with better quality. To reach these requirements it is essential to develop special purpose machine by combining No. of operations. During designing special purpose drilling machine, spindle is the main mechanical component. The main objective of current research work is to design of spindle for special purpose horizontal and angular drilling machine. The spindle is designed by considering bending, torsion and axial load. The spindle is designed for drilling deep hole in cylinder block of cast iron material. After designing the spindle 3D model was created by using the CATIA software. Further, study continues for analysis of spindle by using Finite Element Analysis (FEA) software. Analysis is carried out in static condition. Results of von Mises stresses and displacement from software are compared with theoretical values. Finally, the spindle is checked either it is on safer side or not.

Keywords: *Spindle design, CATIA, FEA, static analysis*

INTRODUCTION

In manufacturing variety of mechanical components, metal cutting operations such as drilling, milling, turning etc. are mostly used [1]. In that, drilling operation is important because more than 60% of all machining processes are related to drilling operation. During designing the spindle, structural properties like dimensions of shaft, motor, tool holding devices, bearings, availability of space, power transmission should be taken into consideration [2]. Also, the trend toward increased productivity and toward better quality with closer tolerances will cause spindle design and analysis to become more important. Vibration in spindle directly affects the surface finish of hole; hence in drilling operation spindle is critical component [3].

Two deep holes are to be drilled in cylinder block of CI FG260 material. One horizontal drill is having 178 mm depth and of diameter 14.25 mm and another angular drill at 54° to entry face with 128.89 mm depth and 14.25 mm hole diameter. In present system of manufacturing process two holes are drilled on separate machine. Horizontal drilling operation is carried out on horizontal machining center and vertical drilling operation is carried on radial drilling machine with indexing jig and fixture at 54°. So, it is required to design a special purpose drilling machine which will perform both operations at a time for improvement in productivity [4].

As both holes are on opposite face of cylinder block, it is required to design separate spindle for each drilling hole and also separate motor should be installed for power requirements. Drilling force of each drill comes directly on the spindle face through drill holder. This thrust force is taken for design consideration. Whole spindle is fixed in housing by using bearings. Total four bearings are mounted. One bearing at drill side of spindle, second at middle portion of spindle and two bearings are mounted back to back to the pulley side of spindle. Motor is mounted on spindle housing by using M. S. plate. As the centre distance in between motor shaft and spindle shaft is less, V belt is preferred for power transmission and hence V pulley is selected. The pulley upward force is taken into consideration for bending moment calculations.

After designing the spindle its 3D model is imported into solid works software and its finite element analysis is done with static condition [5]. For static analysis some assumption are made. The front face of spindle is fixed and torsional moment is given to the spindle from pulley side. The drill's thrust force is directly given to the spindle nose. Third force of bending is applied to the spindle at pulley location in upward direction. In solid works software, the standard bearings are imported at specified location so, that spindle can rotate freely inside the outer races of bearing. In providing conditions of boundary, the outer races of bearing are fixed and inner races of bearing are allowed to freely rotate with spindle. After giving all boundary conditions and constraining all degrees of freedom solid meshing of spindle is done. After meshing, the spindle is studied for maximum Von Mises stresses and displacements developed in spindle [6].

DESIGN OF SPINDLE SHAFT

Power and Force Calculation [7]

For designing any spindle first step is to calculate force exerted by the drill tool.

Work piece material is cast iron FG 260.

Drill diameter = 14.25 mm

Selecting cutting speed = V = 23 m/min

And feed per revolution(s) =0.20 mm/rev

Revolutions per minute

N = (V × 1000) / (π × D) = 513.76 rpm

Selecting material factor = 2.0

Power at spindle = P_s (KW)

$$P_s = \frac{1.25 \times D^2 \times K \times N\,(0.056 + 1.5\,S)}{10^5} = 0.927 \text{ KW}$$

Assume, efficiency of power transmission from motor to spindle (E) = 80%

Power at the Motor = $\frac{P_s}{E}$ = 1158.9 watt

Selecting, Three phase Induction Motor of 2 HP and Speed = 946 rpm.

Thrust force on spindle is calculated by,

$T_h = 1.16 \times k \times D \times (100 \times S)^{0.85}$

= 4138.54 N

Design of Belt and Pulley [8]

Motor is mounted above spindle housing hence distance between spindle shaft and motor shaft is less, so it is preferable to select V belt and V pulley for power transmission. From the speed of faster pulley and designed power, selected belt is of A type. Corresponding values of belt and pulley are calculated as below,

Speed of larger Pulley = 610 rpm

Smaller pulley diameter = 118 mm

Larger pulley diameter = 180 mm

Length of Belt = 1100 mm

Centre distance = 315 mm

Number of V belts = 2

Weight of pulley = 70 N

Calculation of Belt Tensions [9]

During power transmission, belt tension acts radial load on spindle shaft. So, bending moment due to belt tension should be considered during design of shaft. Corresponding values required for calculating tension in belt are as below,

Mass of belt = 0.1144 kg/m

Velocity of belt = 5.844 m/s

Centrifugal tension in belt = 3.907 N

Maximum tension in belt = 260 N

Tension in tight side = 256.093 N

Angle of contact of belt over pulley = 2.944 rad.

Tension in slack side = 20.59 N

Calculation of Bending Moment

The driven pulley is overhang to the extent of 105 mm from the nearest bearing centre. The layout of shaft and bending moment diagram are shown in Figure 1.

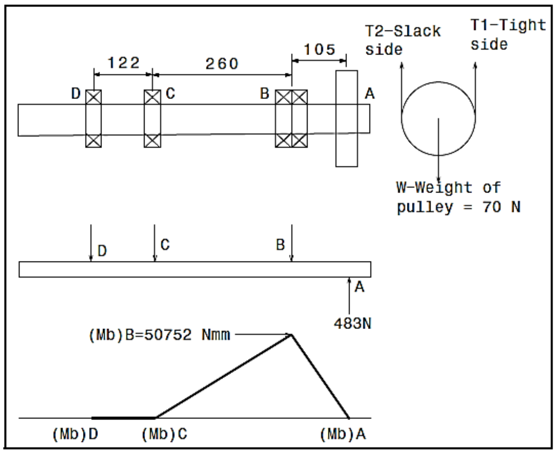

Fig. 1: *Bending Moment Diagram.*

Therefore, bending moment on the shaft due to the belt tensions is calculated as below,

The total upward force acting at the centre line of the pulley is given by,

For Single belt

$F_1 = T_1 + T_2 = 256.09 + 20.59 = 276.68$ N

Total two V belts are used so, total upward force for two belts,

$F_2 = F_1 \times 2 = 553.36$ N

Weight of the pulley is 70 N which is acting downward. Thus, resultant force acting on spindle in upward direction at pulley area = $F_2 - W_1 = 483.36$ N

Bending moment at B is given as,

$(M_b)_B = (F_2 - W_1)\, 105 = 50752$ N-mm

Calculation of Torsional Moment

Torsional Moment = M_t

$$M_t = \frac{\text{Designed Power} \times 60}{2 \times \pi \times N}$$

= 22432 N-mm

Design of Shaft Diameter on Strength Basis [10]

The spindle shaft is designed by considering axial, bending and torsional load. When the shaft is subjected to an axial load in addition to torsion and bending loads, then the stress due to axial load must be added to the bending stress. Shaft is made of ductile material; hence here Maximum shear stress theory is applied for design of shaft.

Notations

K = Radius of gyration

α = Column factor

$\frac{L}{K}$ = Slenderness ratio

K_b = Combined shock and fatigue factor applied to bending moment

K_t = Combined shock and fatigue factor applied to torsional moment

d_i = Inner diameter of shaft

d_o = Outer diameter of shaft

C = Ratio of inside diameter to outside diameter

τ_{max} = Maximum permissible shear stress

F.S. = Factor of safety

F = Thrust force on spindle from drill = 4138.54 N

Calculation of maximum Shear stress

Shaft material is SAE 8620, having

Ultimate tensile strength = 660 N/ mm^2

Yield Tensile strength = 385 N/ mm2

$$\tau_{max} = \frac{0.5 \times Syt}{FS} = 64.16 \text{ N/ mm}^2$$

Pulley is keyed on the shaft hence,

τ_{max} = 0.75 × 64.16 = 48.12 N/ mm2

Assume, $C = \frac{d_i}{do} = 0.6$

Radius of gyration of the hollow shaft is,

$$K = \frac{d_o}{4}\sqrt{1 + \left(\frac{d_i}{d_o}\right)^2}$$

$$K = \frac{d_o}{4}\sqrt{1 + 0.6^2}$$

$$K = 0.2915\, d_o$$

The maximum length in between two bearings is = 228 mm

Thus, Slenderness ratio is $\frac{L}{K} < 115$

So, using following relation, for column factor

$$\alpha = \frac{1}{1 - 0.0044\left(\frac{L}{K}\right)} \quad = \frac{d_o}{d_o - 3.291}$$

Calculate equivalent twisting moment

$$T_e = \frac{\pi}{16} \times \tau_{max} \times (d_o)^3 \times (1 - C^4) \text{….Eq. (1)}$$

$$=8.2238(d_o)^3$$

Equivalent twisting moment also can be calculated by using Torsional, bending moment and axial force as below.

According to ASME code shaft design, the bending and Torsional moments should be multiplied by factors K_b and K_t respectively, to consider for shock and fatigue in shaft during operating condition.

Hence, T_e =

$$\sqrt{[(K_b \times M_b) + \frac{\alpha \times F \times d_o \times (1 + C^2)}{8}]^2 + (K_t \times M_t)^2}$$

…………Eq. (2)

Equating Eq. 1 and Eq. 2 of equivalent twisting moment and from trial and error method, and considering available diameter of drill adapter which fits into inner diameter of spindle.

We select d_o = 55 mm

d_i = 32 mm

d_1 = Shaft diameter where pulley fits

= 50 mm

Selection of Bearing [7]

Radial load is exerted by pulley and axial load is exerted by drill on the spindle during drilling operation. These radial and axial loads are transferred on bearing through spindle. Thus, it is suitable to select bearing such that it carry axial as well as radial load. Hence selecting angular contact ball bearing. For selecting bearing following forces are taken,

F_r = Pure radial load = 553.36 N

F_a = Thrust load = 4138.54 N

C_o = Static load carrying capacity

C = Dynamic load carrying capacity

W = Width in mm

ID = Inner diameter of bearing in mm

OD = Outer diameter of bearing in mm

Life in per million revolutions (L_{10}),

$$L_{10} = \frac{60 \times N \times L_{10h}}{10^6}$$

=1231.2 Million rev

By trial and error method,

C_o = 68081 N

Calculating equivalent dynamic load (P)

$P = (X \times F_r) + (Y \times F_a)$

$P = 5623.58$ N

Dynamic load carrying capacity

$C = P \times L^{1/3} = 60269.68$ N

Selected bearing with its specification are shown in Table1.

Table 1: *Bearing Specification.*

Bearing Designation	ID mm	OD mm	C (N)	C_o (N)	W mm
2 × 7211 CG	55	100	66747	68081	21
2 × 7212 CG	60	110	80098	82767	22

Mounting of Bearings [11]

For assembly point of view 2 × 7211 CG bearings are mounted at B and 2 × 7212 CG bearings are mounted at C and D as shown in Figure 1. Two Bearings with designation 7211 CG are fitted back to back in housing to support the spindle at B [12]. Third Bearing 7212 CG is mounted at D. The length of shaft in between point B and D is more so, mounting forth bearing in between two supports at point C.

ANALYSIS OF SPINDLE

Finite element analysis of the spindle is carried out in following three steps,

Step 1: Preprocessing

Step 2: Processing

Step 3: Post processing

Step 1: Preprocessing

Spindle 3D Modeling

A three dimensional model of spindle is created by using CATIA V5R20, shown in Figure 2.

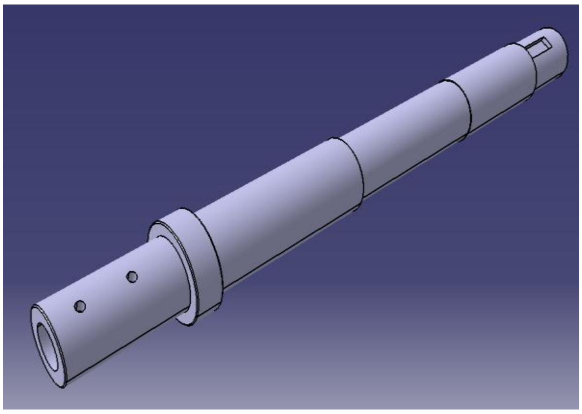

Fig. 2: *Spindle 3D Model.*

Static Analysis

The spindle is analyzed by using solid works software with static condition. In static analysis the effect of steady loading on a structure is considered while inertia and damping effects, such as those caused by time varying loads are ignored [13]. Static analysis can include steady inertia loads such as gravity and rotational velocity and time varying loads can be approximated as equivalent to static load. The static analysis is used to determine the, von Mises stresses, displacements, strains by applying various forces in structures or components.

Importing Bearings

The spindle 3D model is imported in solid works software from CATIA. Standard bearings with required inner and outer diameter and having dynamic and static load carrying capacity as per designed calculation are selected and mounted at specified location on the spindle. The proper bearings are mounted on spindle by using solid works software as shown in the Figure 3.

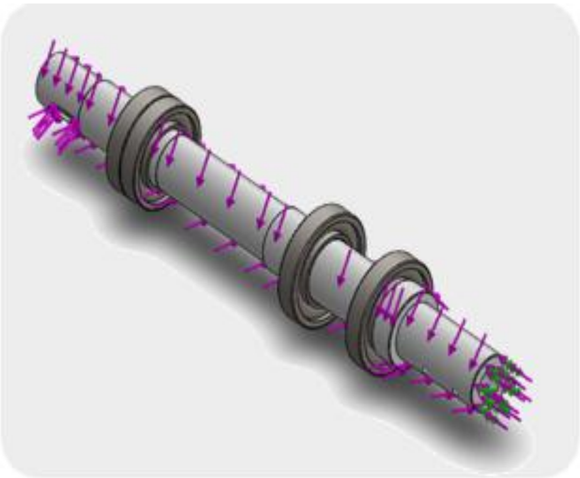

Fig. 3: *Bearing Locations Along with Spindle.*

Properties of Spindle Material

SAE 8620 material was used for spindle alloy steel with following mechanical properties shown in Table 2.

Table 2: *Properties of SAE 8620 Material.*

Property	**Value**
Density	7859kg/m^3
Tensile Strength	660 N/mm^2
Yield Strength	385 N/mm^2
Young's Modulus	2.1×10^{11} N/m^2
Poisson Ratio	0.29
Brinell Hardness	149 HB
Elongation	31%
Machinability	65%
Shear Modulus	80×10^9 N/m^2
Thermal Expansion Coefficient	42 /Kelvin

Boundary Conditions

During working condition of spindle the bearings inner races are fixed with spindle and outer races are fixed with housing. Hence, in providing boundary conditions for analysis purpose, all of the outer races of bearing are fixed and inner races are allowed to freely rotate with spindle. One more assumption is considered in a case that drill is jammed in hole and stability of spindle is studied against failure. Drill holder is tightened on the front face of spindle, hence when drill is jammed, in case of spindle its front face can be considered as fixed. Hence finally, spindle is fixed at five faces that are at four bearings outer races and fifth at spindle's front face, as shown in Figure 4 with blue color.

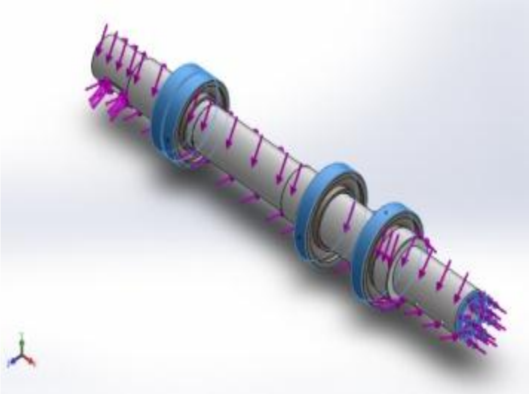

Fig. 4: *Boundary Conditions at Five Faces.*

Application of Different Loads

Drilling thrust force is directly coming on the front face of spindle through drill adapter. Calculated thrust force is equal to 4138.54 N is applied on the front face of spindle as shown in Figure 5. Second force is due to belt tension applied in upward direction at pulley and key location. This force is applied in radial direction to the spindle with 483.36 N as shown in Figure 6.

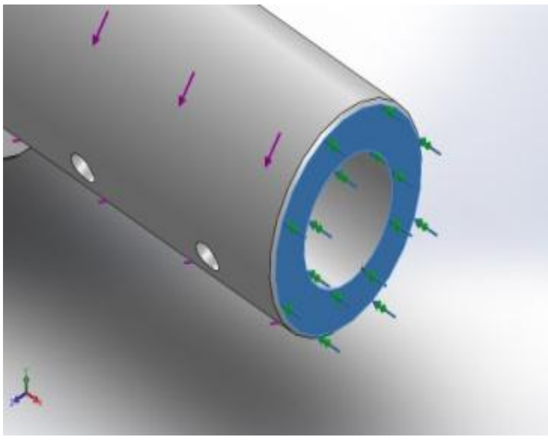

Fig. 5: *Thrust Force Applied on Front Face.*

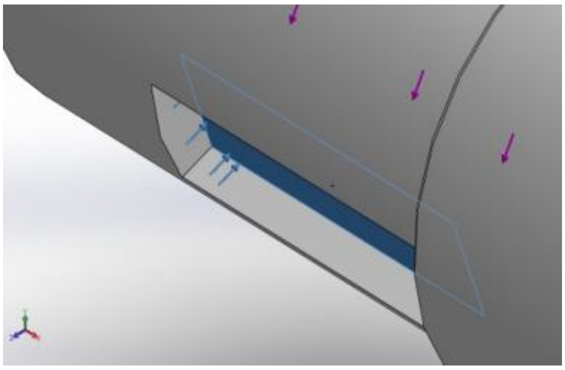

Fig. 6: *Radial Force Applied at Key Way in Upward Direction.*

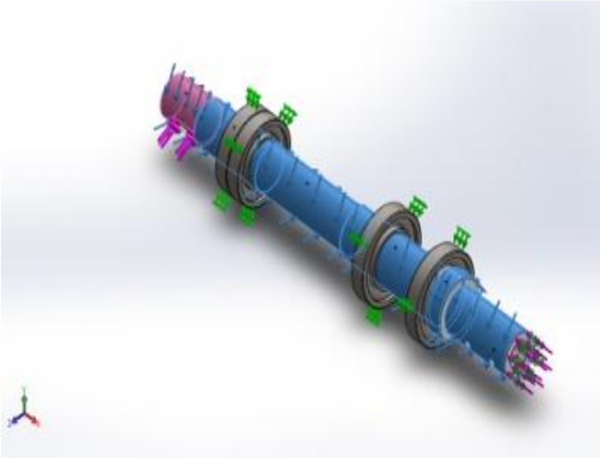

Fig. 7: *Torsional Moment Applied Throughout Shaft.*

Third force is torsional moment force and is applied throughout spindle shaft in clockwise direction when viewed from rear side as shown in Figure 7. Applied value of torsional moment is equal to 22.4 N.m.

Meshing of Spindle

Tetrahedral element is used for meshing a spindle. Average size of element used for meshing is 7.34427 mm with tolerance of

0.367213. Tetrahedral element meshing is as shown in Figure 8.

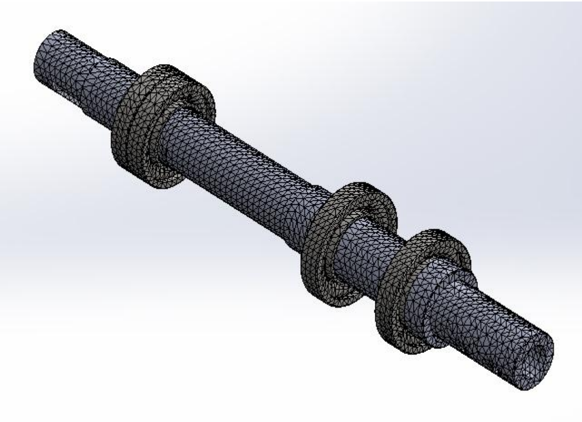

Fig. 8: *Meshed Model.*

Step 2: Processing

After uploading complete data of the spindle in solid works software, it is solved for solution. Internally software carries out matrix formations, inversion, multiplication and gives solution for displacement and stress.

Step 3: Post Processing

Results and Discussion

Stress Analysis

After meshing, stress analysis is done for the spindle. The maximum value of von Mises stress is reached to 9.25×10^6 N/m^2 and minimum value of von Mises stress is 4.087×10^{-3} N/m^2. According to distortion energy theory the ductile solid material yields when the von Mises stress exceed the yield value exceeds the yield stress value of the materials [14]. The yield tensile strength of the spindle material Alloy steel SAE 8620 is 3.85×10^8 N/m^2. The von Mises stress is less than the yield strength; hence design of spindle is on safer side [15]. Von Mises stresses at different node point on the surface of spindle are as shown in Figure 9. Stress analysis shows that maximum Von Mises stress occurs near spindle nose and near rear bearing support due to thrust force and belt tension respectively. The von Mises stress value at node point 47983 is 6.47×10^6 N/m^2, which is larger as compare to other spans of spindle, due to belt tension in upward direction. The distance between rear bearing support and middle bearing support is quite large than front and middle bearing support. Hence, stress in between rear and middle bearing support is greater than other two bearing support. The stress at node point 34627 in between two bearing support is 2.101×10^6 N/m^2. The stress at a node point 2957 near to the spindle nose is 3.861×10^6 N/m^2. From the values it is clear that stress due to bending is greater than stress due to thrust force.

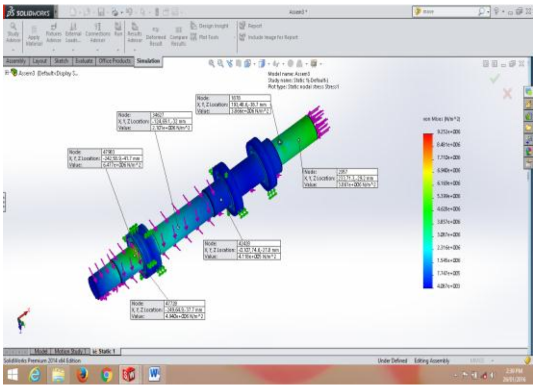

Fig. 9: *Stress Analysis of Spindle.*

Displacement Analysis

Further, study is continued for analysis of resultant displacement of the same spindle. Maximum displacement of shaft should not exceed 2×10^{-4} times the span between bearings [7]. Thus, Maximum theoretical value of displacement for span AB (Figure 1) having length 105 mm will be 21×10^{-3} mm. The displacement analysis of the spindle is shown in Figure 10. Displacement at various nodes is represented in Table 3.

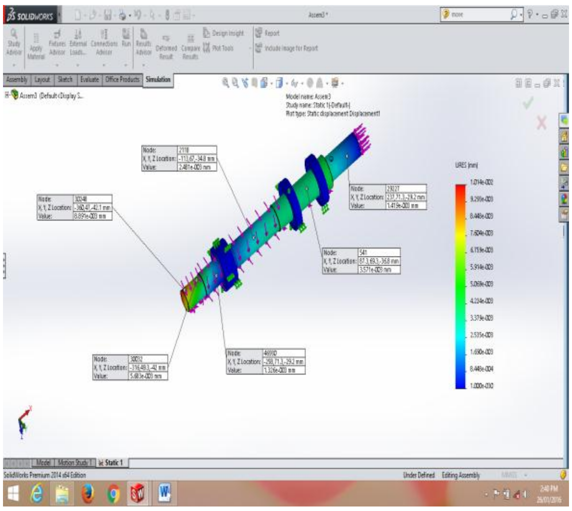

Fig. 10: *Displacement Analysis of Spindle.*

Table 3: *Displacement at Different Nodes.*

Node Point	Displacement mm
29227	1.41×10^{-3}
541	3.571×10^{-3}
2118	2.481×10^{-3}
46930	1.32×10^{-3}
30248	8.91×10^{-3}
37627	10.138×10^{-3}
30032	5.683×10^{-3}

The maximum displacement occurred for spindle is 10.138×10^{-3} mm at node point 37627 in between span AB which is due to the belt tension. And the theoretical value of maximum displacement for span AB is 21×10^{-3} mm. Displacement value by software is less than theoretical vale of maximum displacement hence, the spindle design is safe.

CONCLUSION

In current research work, design and analysis of spindle for special purpose drilling machine was carried out which is used for drilling hole of Φ14.25 mm with 178 mm depth. After designing, 3D model of spindle was created in CATIA V5R20 software and finite element analysis was carried out using solid works software in static condition. The obtained value of von Mises stress is 9.2516×10^{6} N/m^2 which is less than yield tensile strength of the spindle material. Also, maximum displacement in the spindle is less than theoretical value of maximum permissible displacement. Hence, design is safe.

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